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INFLUENCE OF INJECTION TIMING ON PERFORMANCE PARAMETERS AND COMBUSTION CHARACTERISTICS OF HIGH GRADE SEMI ADIABATIC DIESEL ENGINE WITH COTTON SEED BIODIESEL

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ABSTRACT

Biodiesel obtained from feed stocks of vegetable oils are important substitutes of diesel fuel, as they are renewable and biodegradable. However, with moderate viscosity and low volatility of biodiesel cause combustion problems in CI engines, call for hot combustion chamber engines, which provide hot environment for burning moderate viscous biodiesel fuels. With cotton seed biodiesel investigations were carried out to evaluate the performance of low heat rejection combustion chamber. It consisted of an air gap insulated piston with superni (an alloyof nickel) crown, an air gap insulated liner with superni insert and ceramic coated cylinder head with cotton seed biodiesel with varied injection timing. Performance parameters of brake thermal efficiency, brake specific energy consumption, exhaust gas temperature, coolant load and volumetric efficiency were determined at full load operation of the engine. At full load operation of the engine Combustion characteristics of peak pressure, maximum rate of pressure rise and time of occurrence of peak pressure were evaluated by means of Piezo electric transducer, TDC encoder and special pressure-crank angle software package. The optimum injection timing for conventional engine (CE) and LHR combustion chamber were 31^o bTDC and 28^o bTDC (before top dead centre) respectively with biodiesel. Comparative studies were made at manufacturer's recommended injection timing (27^{o}bTDC) and optimum injection timing with biodiesel operation for CE and engine with LHR combustion chamber. Engine with LHR combustion chamber with biodiesel showed improved performance and combustion characteristics at 27^{o}bTDCand at optimum injection timing over CE.

KEYWORDS: Vegetable Oil, Biodiesel, Conventional Engine, LHR Combustion Chamber, Fuel Performance, Combustion characteristics, Injection Timing

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Original Article

INTRODUCTION

Fossil fuels are limited resources for ensuring energy security and environmental protection; hence, search for renewable fuels is becoming more and more prominent. It has been found that the vegetable oils are promising substitute for diesel fuel, because of their properties are comparable to those of diesel fuel, renewable and can be easily produced. A century ago Rudolph Diesel, first who invented the diesel engine, demonstrated the principle by employing peanut oil and hinted that vegetable oil would be the future fuel in diesel engine [1].Several researchers experimented the use of vegetable oils as fuel on conventional engines (CE) and reported that the performance was poor, citing the problems of high viscosity, low volatility and their polyunsaturated character. The use of vegetable oils caused the problems of piston ring sticking, injector and combustion chamber deposits, fuel system deposits,

reduced power, reduced fuel economy and increased exhaust emissions [1–5].

The crude vegetable oils can be chemically modified (esterified) to biodiesel to minimize the problems of crude vegetable. Biodiesels were produced from vegetable oils present a very promising alternative for diesel fuel, as the biodiesels have numerous advantages compared to fossil fuels. They are renewable, biodegradable, provide energy security and foreign exchange savings besides addressing environmental concerns and socio–economic issues. Investigations were made with biodiesel on CE[6–10].They reported from their studies that biodiesel operation showed comparable thermal efficiency, reduced particulate emissions and increased nitrogen oxide (NO_x) levels, when compared with mineral diesel operation.

Experiments were conducted and observed the problem of injection, on preheated vegetable oils in order to equalize their viscosity to that of mineral diesel may ease the problems of injection process [11–13]. Experiments were carried out with preheated vegetable oils on engine and compared with normal biodiesel. They reported that engine with preheated vegetable oils marginally improved thermal efficiency, reduced particulate matter emissions and NO_x levels, than engine with normal diesel.

Increased injector opening pressure may also result significant efficient in combustion of compression ignition engine [14–15]. It effects on performance and formation of pollutants inside the direct injection diesel engine combustion. Studies were made with increase of injector opening pressure on engine with biodiesel. They reported that performance of the engine was improved, particulate emissions were reduced and NO_x levels were increased marginally with an increase of injector opening pressure.

The limitations associated with biodiesel (high viscosity and low volatility) call for hot combustion chamber, provided by low heat rejection (LHR) combustion chamber. The importance of the engine with LHR combustion chamber is reduce heat loss to the coolant with provision of thermal barrier in the path of heat flow to the coolant. Three different approaches that are being pursued to decrease heat rejection are (1) Coating with low thermal conductivity materials on cylinder head, crown of the piston and inner portion of the liner (low grade LHR combustion chamber); (2) air gap insulation where air gap is provided in the piston and other components with low-thermal conductivity materials like superni (an alloy of nickel),cast iron and mild steel(medium grade LHR combustion chamber);and (3)high grade LHR engine is a combination of air gap insulation and ceramic coated components.

Experiments were conducted with biodiesel, an air gap (3 mm) insulation in piston as well as in liner and ceramic coated cylinder head on engine with high grade LHR combustion chamber. The engine was fuelled with biodiesel with varied injector opening pressure and injection timing [16–22]. They reported from their investigations, that engine with high grade LHR combustion chamber at an optimum injection timing of 28^{o}bTDC with biodiesel increased brake thermal efficiency by 10–12%, at full load operation–decreased particulate emissions by 45–50% and increased NO_x levels, by 45–50% when compared with mineral diesel operation on CE at 27^o bTDC.

The present paper attempted to determine the performance of the engine with high grade LHR combustion chamber. It consist of an air gap (3.2 mm) insulated piston, an air gap (3.2 mm) insulated liner and ceramic coated cylinder head with cotton seed biodiesel with varied injection timing. Results were compared with CE with biodiesel and also CE with diesel at similar operating conditions.

MATERIALS AND METHODS

The oil content of cotton seeds has approximately 18% (w/w). Cottonseed production in India is estimated to be around 35%of its cotton output (approximately 4.5millionmetric tons). Cottonseed oil produced in India is approximately 0.30 million metric ton and it is an attractive biodiesel feedstock [8]

Preparation of Biodiesel

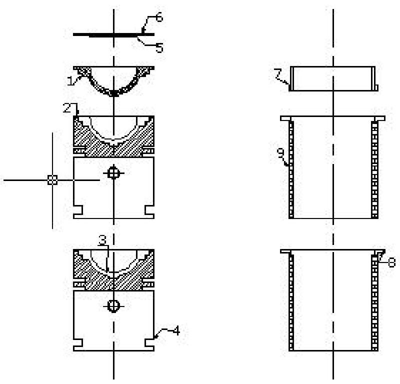
The preparation of biodiesel was mentioned in Ref [25].The properties of the Test Fuels used in the experiment were presented in Table 1. [8].

Table 1: Properties of Test Fuels [8]

Property	Units	Diesel (DF)	Biodiesel(BD)	ASTM Standard
Carbon Chain	--	C_8–C_{28}	C_{16}–C_{24}	---
Cetane Number	-	51	56	ASTM D 613
Specific Gravity at 15°C	-	0.8275	0.8673	ASTM D 4809
Bulk Modulus at 15°C	MPa	1408.3	1564	ASTM D 6793
Kinematic Viscosity @ 40°C	cSt	2.5	5.44	ASTM D 445
Air Fuel Ratio (Stoichiometric)	--	14.86	13.8	--
Flash Point (Pensky Marten's Closed Cup)	°C	120	144	ASTM D93
Cold Filter Plugging Point	°C	Winter 6° C Summer 18°C	3° C	ASTM D 6371
Pour Point	°C	Winter 3°C Summer 15°C	0°C	ASTM D 97
Sulfur	(mg/kg, max)	50	42	ASTM D5453
Low Calorific Value	MJ/kg	42.0	39.9	ASTM D 7314
Oxygen Content	%	0.3	11	--

Engine with LHR Combustion Chamber

Figure 1 shows assembly details of insulated piston, insulated liner and ceramic coated cylinder head. Engine with LHR combustion chamber contained a two–part piston; the top crown made of superni was screwed to aluminium body of the piston, providing an air gap (3.2 mm) by placing a superni gasket in between the body and crown of the piston.



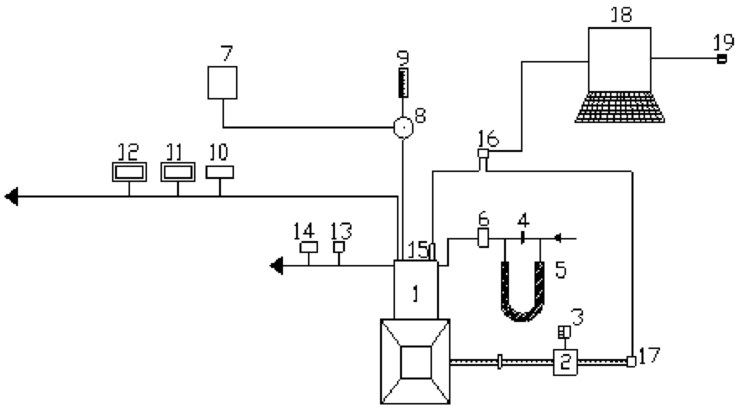
1. Pistoncrown with threads 2. Superni gasket 3.Air gap in piston 4. Body of piston 5. Ceramic coating on inside portion of cylinder head 6. Cylinder head, 7.Superni insert with threads, 8.Air gap in liner, 9. Liner

Figure 1: Assembly Details of Air Gap insulated Piston, Air Gap Insulated Liner and Ceramic Coated Cylinder Head

An air gap of 3.2 mm was maintained between the liner body and the superni insert, as superni insert was screwed to the top portion of the liner. At 500 ^oC the thermal conductivity of superni and air are 20.92 and 0.057 W/m–K. Partially stabilized zirconium (PSZ) of thickness 500 microns was coated by means of plasma coating technique as a thermal barrier. The combination of air, low thermal conductivity materials of superni and PSZ provide sufficient insulation for propagation of heat to the coolant, thus resulting in LHR combustion chamber.

Experimental Set–up

The schematic diagram shown in Figure 2 illustrate the experimental setup used for the investigations on the engine with LHR combustion chamber with cotton seed biodiesel. Table 2 about specifications of Test engine. The engine was coupled with an electric dynamometer (Kirloskar) and loaded by a loading rheostat. Burette was used to measure fuel rate. The accuracy of brake thermal efficiency obtained is ±2%. Provision for preheating of biodiesel to the required levels (90^oC) was made to equalize the viscosity of biodiesel to that of diesel fuel at room temperature. Air box method was used to measure air-consumption of the engine with air box, orifice flow meter and U–tube water manometer assembly. The naturally aspirated engine with water–cooling system was provided in which outlet temperature of water was maintained at 80^oC by adjusting the flow rate of water. The analogue water flow meter with an accuracy measurement of ±1%was used to measure the flow rate of water.



1. Four Stroke Kirloskar Diesel Engine 2. Kirloskar Electrical Dynamometer 3.Load Box 4.Orifice flow meter 5.U-tube water manometer, 6.Air box 7.Fuel tank 8. Pre-heater 9.Burette 10. Exhaust gas temperature indicator 11.AVL Smoke opacity meter 12. Netel Chromatograph NO_xAnalyzer 13.Outlet jacket water temperature indicator 14. Outlet-jacket water flow meter 15.AVL Austria Piezo-electric pressure transducer 16.Console 17.AVL Austria TDC encoder 18.Personal Computer and 19.Printer.

Figure 2: Schematic Diagram of Experimental Set–up

Engine oil was provided with a pressure feed system. To measure the lube oil temperature no temperature control were incorporated. The injection timing was varied by incorporating copper shims of suitable size provided in between the pump body and the engine frame. Using nozzle testing device Injector opening pressure was changed from 190 bar to 270 bar.s

Table 2: Specifications of Test Engine

Description	Specification
Engine make and model	Kirloskar (India) AV1
Maximum power output at a speed of 1500 rpm	3.68 kW
Number of cylinders ×cylinder position× stroke	One × Vertical position × four-stroke
Bore × stroke	80 mm × 110 mm
Engine Displacement	553 cc
Method of cooling	Water cooled
Rated speed (constant)	1500 rpm
Fuel injection system	In-line and direct injection
Compression ratio	16:1
BMEP @ 1500 rpm at full load	5.31 bar
Manufacturer's recommended injection timing and injector opening pressure	27^{o}bTDC × 190 bar
Number of holes of injector and size	Three × 0.25 mm
Type of combustion chamber	Direct injection type

The maximum injector opening pressure was restricted to 270 bar due to practical difficulties involved. Iron and iron-constantan thermocouples were connected to analogue temperature indicators to measure the coolant water jacket inlet temperature, outlet water jacket temperature and exhaust gas temperature. The accuracies of analogue temperature indicators are ±1%. Piezo electric transducer, (AVL Austria: QC34D) connected to cylinder head. TDC encoder (AVL Austria: 365x) connected to extended portion of dynamometer shaft, which were connected to console. Console was connected to compute. Crank angle diagram was obtained from the signals of pressure and crank angle. The accuracy of measurement of pressure and crank angle were ±1 bar, and ± 1^{o} respectively. Combustion parameters such as peak pressure (PP), time of occurrence of peak pressure (TOPP) and maximum rate of pressure rise at the full load operation of the engine were evaluated.

RESULTS AND DISCUSSIONS

The optimum injection timing were 31^{o}bTDC and 28^{o}bTDC for CE and engine with LHR combustion chamber with diesel operation respectively [23-24]. Similarly the optimum injection timing for biodiesel operation were 31^{o} bTDC and 28^{o} bTDC. [25].

Figure 3 presents bar charts showing the variation of peak BTE with test fuels. From Figure 3, it is observed that engine with LHR combustion chamber operated with biodiesel raised peak BTE by 3% at 27^{o} bTDC and 10% at 31^{o} bTDC in comparison with CE with biodiesel operation at 27^{o} bTDC and at 31^{o} bTDC. Hot environment of the engine with LHR combustion chamber improved evaporation of biodiesel, might have improved peak brake thermal efficiency of the engine. With test fuels and same configuration of the engine, engine with LHR combustion chamber with biodiesel operation showed higher peak BTE than with operation of diesel. From Peak BTE observed that engine with LHR combustion chamber was more suitable for biodiesel.

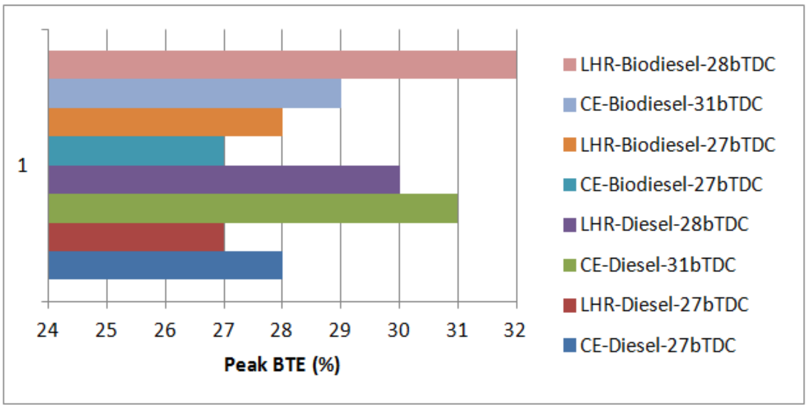


Figure 3

Figure 3 Bar charts showing the variation of peak brake thermal efficiency (BTE) with test fuels with conventional engine (CE) and engine with LHR combustion chamber at recommended and optimized injection timings at an injector opening pressure of 190 bar.

Figure 4 presents bar charts showing the variation of brake specific energy consumption (BSEC) at full load with test fuels. From Figure 4, it is shown that with advanced injection timing with test fuels BSEC at full load operation decreased. This was due to increase of resident time of fuel with air thus improving atomization and thus combustion. BSEC was comparable with biodiesel with CE at 27^{o} bTDC and 31^{o}bTDC, when compared with CE with diesel operation at 27^{o} bTDC and at 31^{o}bTDC. Improved combustion with higher cetane number and presence of oxygen in fuel composition with higher heat release rate with biodiesel may lead to produce comparable BSEC at full load. BSEC lowered at full load operation with biodiesel, with engine with LHR combustion chamber by 6% at 27^{o} bTDC and 3% at 28^{o} bTDC, when compared diesel operation with engine with LHR combustion chamber at 27^{o} bTDC and at 28^{o} bTDC. This once again observed that LHR engine was more suitable for biodiesel operation than neat diesel operation. At full load operation biodiesel as test fuel LHR engine decreased BSEC by 3% at 27^{o}bTDC and 2% at 28^{o} bTDC, in comparison with CE at 27^{o} bTDC and at 31^{o} bTDC with biodiesel. The performance of LHR engine might have improved with improved rate of evaporation and higher heat release rate of fuel with LHR combustion chamber.

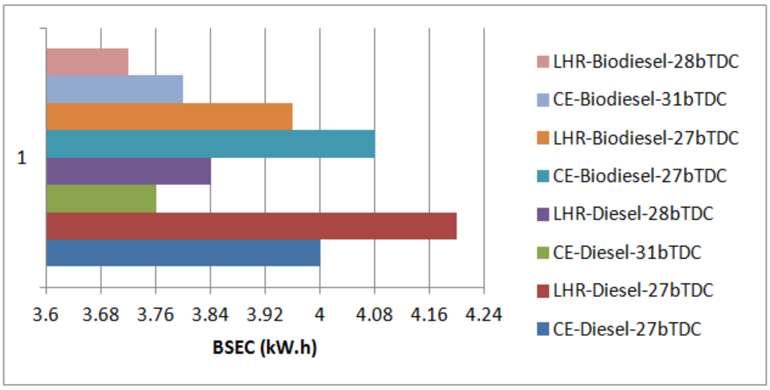


Figure 4

Figure 4 Bar charts showing the variation of brake specific energy consumption (BSEC) at full load operation with test fuels with both versions of the engine at recommended and optimized injection timings at an injector opening pressure of 190 bar.

Figure 5 presents bar charts showing variation of exhaust gas temperature (EGT) at full load with test fuels. From Figure 5, it is noticed that, exhaust gas temperature (EGT) at full load operation lowered with advanced injection timing with test fuels as the work transfer from the piston to the gases in the cylinder at the end of the compression stroke was too large. At full load operation CE with biodiesel operation increased EGT by 6% at 27^{o}bTDC and 7% at 31^{o}bTDC in comparison with CE with neat diesel operation at 27^{o}bTDC and at 31^{o}bTDC. The density of biodiesel is higher, therefore higher amount of heat was released in the combustion chamber leading to produce higher EGT at full load operation with biodiesel operation than neat diesel operation even though calorific value (or heat of combustion) of biodiesel is lower than that of diesel. This was also because of higher duration of combustion of biodiesel causing retarded heat release rate. Similar findings were observed by other researchers [6–8]. From Figure 5, it is observed that at full load operation LHR engine with biodiesel operation increased EGT by 5% at 27^o bTDC and 5% at 28^o bTDC, when compared with diesel operation at 27^o bTDC and at 28^o bTDC on same configuration of the engine. High combustion duration due to high viscosity of biodiesel in comparison with diesel might have increased EGT at full load. LHR engine with biodiesel increased EGT at full load operation by 17% at 27^{o}bTDC and 13% at 28^o bTDC, in comparison with CE at 27^{o}bTDC and at 31^o bTDC. This denoted that heat rejection was restricted through the piston, liner and cylinder head, thus maintaining the hot combustion chamber as result of which EGT at full load operation increased with reduction of ignition delay. Similar observations were reported by previous researchers [21–22].

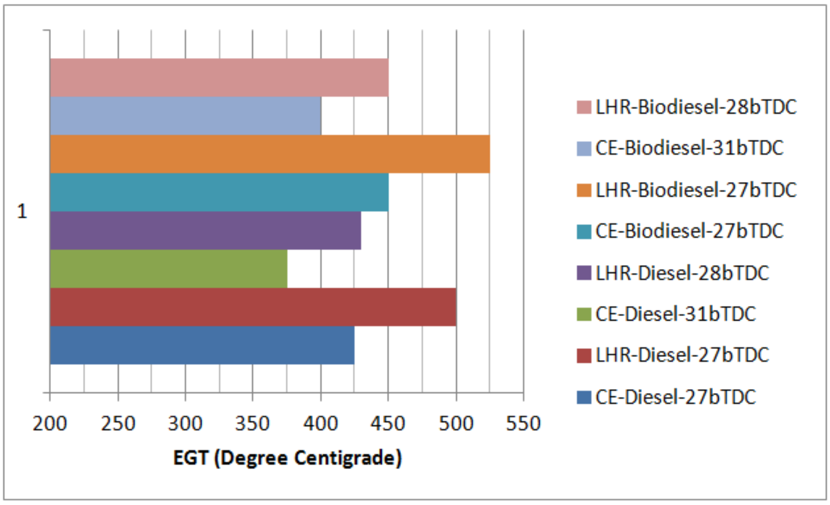


Figure 5

Figure 5 Bar charts showing the variation of exhaust gas temperature (EGT) at full load operation with test fuels with both versions of the engine at recommended and optimized injection timings at an injector opening pressure of 190 bar.

Figure 6 represents bar charts showing variation of coolant load with test fuels. CE with biodiesel operation increased coolant load by 3% at 27^{o}bTDC and 10% at 31^{o}bTDC in comparison with neat diesel operation on CE at 27^o bTDC and 31^o bTDC as observed from Figure 6.

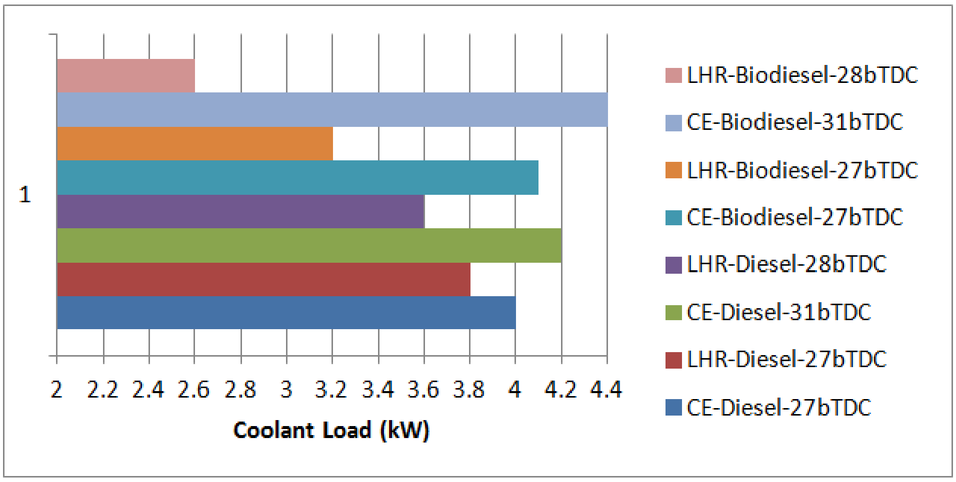


Figure 6: Bar Charts Showing the Variation of Coolant Load at Full Load with Test Fuels with Both Versions of the Engine at Recommended and Optimized Injection Timings at an Injector Opening Pressure of 190 Bar

Presence of un–burnt fuel concentration at the walls of combustion chamber may lead to increase of gas temperatures with biodiesel produced higher coolant load than diesel operation. Similar trends were reported in previous studies [21–22]. Coolant load at full load operation with advanced injection timing with biodiesel increased in CE, while decreasing the same in LHR engine. In case of CE, un–burnt fuel concentration reduced with effective utilization of energy, released from the combustion, coolant load with test fuels increased marginally at full load operation, with an increase of gas temperatures, when the injection timing was advanced to the optimum value. The reduction of coolant load in LHR engine might be due to the reduction of gas temperatures with improved combustion. Hence, the improvement in the performance of CE was due to heat addition at higher temperatures and rejection at lower temperatures, while the improvement in the efficiency of the engine with LHR combustion chamber was because of recovery from coolant load at their optimum injection timings with test fuels. LHR engine with biodiesel operation decreased coolant load operation by 16% at 27^{o} bTDC and 28% at 28^{o} bTDC, when compared diesel operation with same configuration of the engine at 27^{o} bTDC and at 28^{o} bTDC. More conversion of heat into useful work with biodiesel than diesel might have reduced coolant load with biodiesel. Figure 6 indicates that engine with LHR combustion chamber with biodiesel decreased coolant load at full load operation by 7% at 27^{o}bTDC and 41% at 28^{o} bTDC, in comparison with CE at 27^{o}bTDC and at 31^{o} bTDC. Provision of thermal insulation and improved combustion with engine with LHR combustion chamber might have reduced coolant load with LHR engine in comparison with CE with biodiesel operation. Similar observations were reported by previous researchers. [21–22].

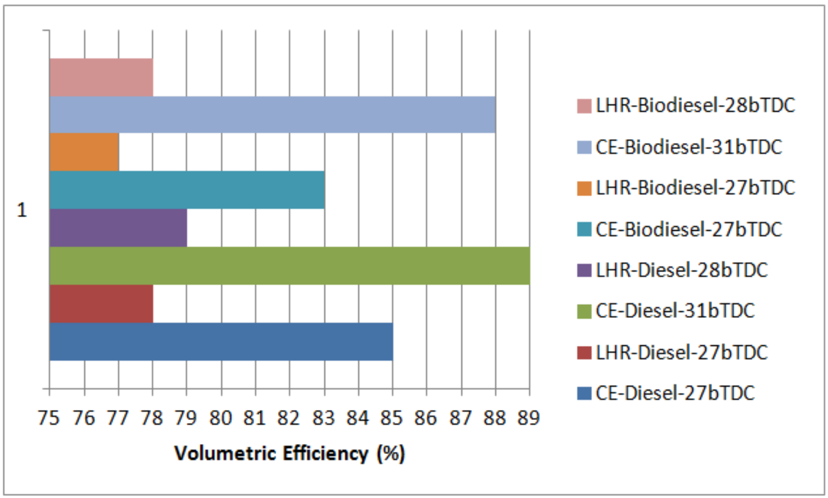


Figure 7: Shows bar Charts Showing Variation of Volumetric Efficiency at Full Load with Test Fuels

Figure 7 Bar charts showing the variation of volumetric efficiency at full load operation with test fuels with both versions of the engine at recommended and optimized injection timings at an injector opening pressure of 190 bar.

It indicates that CE with biodiesel operation decreased volumetric efficiency at full load by 2% at 27^{o}bTDC and comparable at 31^{o}bTDC, when compared with diesel operation on CE at 27^o bTDC and 31^o bTDC. Increase of EGT might have reduced volumetric efficiency at full load, as volumetric efficiency depends on combustion wall temperature, which in turn depends on EGT. Volumetric efficiency at full load operation improved marginally with advanced injection timing with test fuels with both configurations of the combustion chamber. Reduction of EGT at full load might have improved volumetric efficiency with test fuels. From Figure 7, it is noticed that volumetric efficiencies at full load operation on engine with LHR combustion chamber at 27^o bTDC and at 28^o bTDC with biodiesel were marginally lower than diesel operation on same configuration of the engine at 27^o bTDC and 28^o bTDC. Increase of EGT was responsible factor for it. Figure 7indicates that engine with LHR combustion chamber with biodiesel decreased volumetric efficiency at full load operation by 7% at 27^{o}bTDC and 11% at 28^o bTDC, in comparison with CE at 27^{o}bTDC and at 31^o bTDC. The reduction of volumetric efficiency with engine with LHR combustion chamber was because of increase of temperatures of insulated components of LHR combustion chamber, which heat the incoming charge to high temperatures and consequently the mass of air inducted in each cycle was lower. Similar observations were noticed by earlier researchers [21–22].

Combustion Characteristics

Figure 8 represents bar charts showing variation of peak pressure with test fuels. From Figure 8, it is observed that CE with biodiesel at full load operation increased peak pressure (PP) by 4% at 27^{o}bTDC and 5% at 31^o bTDC when compared with diesel operation on CE at27^{o}bTDC and at 31^o bTDC. Even though biodiesel has lower heat of combustion, it advanced its peak pressure position because of its higher bulk modulus and cetane number. This shift is mainly due to advancement of injection due to higher density and earlier combustion due to shorter ignition delay caused by higher cetane number of biodiesel. Peak pressure at full load operation with biodiesel increased with CE, while marginally decreasing it in LHR engine with advanced injection timings. Increase of ignition delay of fuel in CE, causing sudden increase of pressures with accumulated fuel with advanced injection timings increased peak pressure in CE. Improved combustion with improved air–fuel ratios decreased peak pressure at full load operation on engine with LHR combustion chamber. Figure 8 indicates that engine with LHR combustion chamber with biodiesel increased peak pressure at full load

operation by 35% at 27^{o}bTDC and 5% at 28^{o} bTDC, in comparison with CE at 27^{o}bTDC and at 31^{o} bTDC. Improved heat release rate with engine with LHR combustion chamber might have increased peak pressure with LHR engine with biodiesel. Peak pressure at full load operation increased in CE, while marginally reducing it with engine with LHR combustion chamber with an increase of injector opening pressure with test fuels. Smaller sauter mean diameter shorter breakup length, better dispersion, and better spray and atomization characteristics might be responsible factors to increase peak pressure with CE with biodiesel. This improves combustion rate in the premixed combustion phase in CE. Improved combustion with improved air–fuel ratios marginally reduced peak pressure at full load operation with LHR engine.

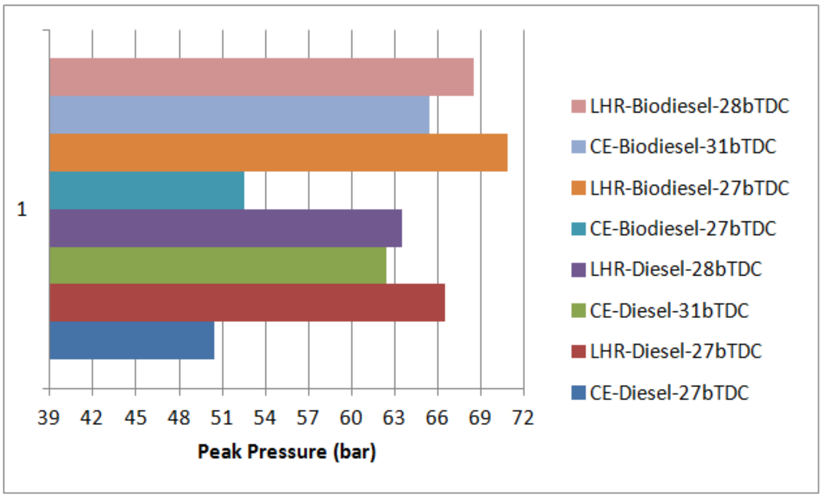


Figure 8: Bar Charts Showing the Variation of Peak Pressure at Full Load Operation with Test Fuels with Both Versions of the Engine at Recommended and Optimized Injection Timings at an Injector Opening Pressure of 190 Bar

Figure 9 presents bar charts showing the variation of maximum rate of pressure rise at full load with test fuels at recommended injection timing and optimum injection timing. Figure 9 represents that maximum rate of pressure rise (MRPR) was higher for diesel than biodiesel with both versions of the combustion.

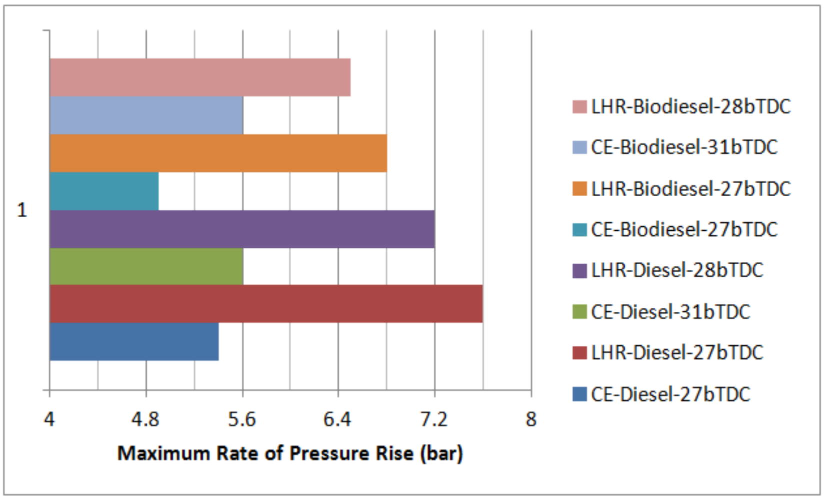


Figure 9: Bar Charts Showing the Variation of Maximum Rate of Pressure Rise (MRPR) at Full Load Operation with Test Fuels with Both Versions of the Engine at Recommended and Optimized Injection Timings at an Injector Opening Pressure of 190 Bar

MRPR at full load might have increased due to High volatile nature of diesel fuel releasing more energy per unit crank angle. MRPR at full load operation followed similar trends at different operating conditions as in case of PP at full load operation. LHR engine increased MRPR at full load operation by 39% at 27^{o}bTDC and 16% at 28^{o}bTDC, when compared with CE at 27^{o}bTDC and at 31^{o}bTDC with biodiesel, which showed that combustion improved with improved air–fuel ratios in hot environment provided by the LHR engine.

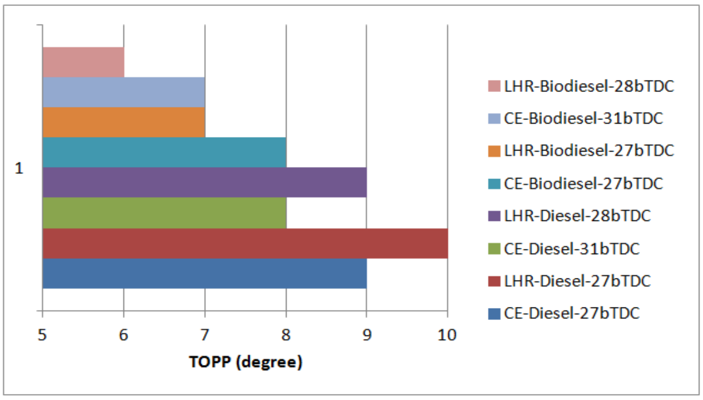


Figure 10: Presents Bar Charts Showing the Variation of Time of Occurrence of Peak Pressure (TOPP) at Full Load with Test Fuels at Recommended Injection Timing and Optimum Injection Timing

Figure 10 Bar charts showing the variation of time of occurrence of peak pressure (TOPP) at full load operation with test fuels with both versions of the engine at recommended and optimized injection timings at an injector opening pressure of 190 bar.

AT recommended injection timing and optimum injection timing with biodiesel operation at full load, marginally decreased time of occurrence of peak pressure (TOPP) on CE, while drastically decreasing it with LHR engine, when compared with diesel operation, as noticed from Table 1. Higher bulk modulus of rigidity and cetane number of biodiesel, when compared with neat diesel operation might have reduced TOPP at full load operation. TOPP at full load operation decreased (towards TDC) with the advanced injection timing with both versions of the combustion chamber as noticed from Figure 10, which showed that combustion improved with advanced injection timing.

SUMMARY

- LHR engine is efficient for alternative fuel like biodiesel rather than neat diesel.
- At recommended injection timing and optimized timing engine with LHR combustion chamber with biodiesel improved its performance over CE.
- With both versions of the combustion chamber with biodiesel with advanced injection timing improved the performance of the engine and combustion parameters of the engine.

Novelty

To improve the performance and combustion characteristics of the engine, engine parameter (injection timing) and different configurations of the engine (conventional engine and engine with LHR combustion chamber) were used simultaneously. Change of injection timing was accomplished by incorporating copper shims between pump frame and engine body.

CONCLUSIONS

- Injection timings of fuel affect engine performance and combustion parameters.
- Change of combustion chamber design from CE to LHR combustion chamber improved the performance and combustion parameters of the engine

Future Scope of Work

The performance and combustion characteristics of the engine can further improved by preheating of the biodiesel and increase of injection pressure.

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